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NUMERICAL AND EXPERIMENTAL ANALYSIS OF A LINEAR COMPRESSOR

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ABSTRACT

The paper presents the study of a new compressor for household appliances, developed through a virtual simulation system, with Simulink and Adams software. Linear compressor has a linear motor coupled to a mechanical system: a piston and a spring. The advantage of this compressor respect to a standard reciprocating compressor is due to the high reduction of energy losses and variable capacity provided. From the numerical optimized model a running prototype compressor has been produced, which obtain the high performances predicted by the virtual model.

NOMENCLATURE

z_1	stator position	b_1	supporting spring damping	K_1	supporting spring stiffness
m_1	stator mass	z_2	piston position	K_2	piston spring stiffness
b_2	piston spring damping	m_2	piston mass		
F_{el}	electric force	F_{th}	pressure force	F_T	total force
i	current	B	magnetic flux density	B_{eff}	effective flux density
l_{eff}	effective length wire	V	voltage source	V_1	motion induced tension
Φ	magnetic flux	L_{eq}	equivalent inductance	R_i	iron losses
L_c	coil inductance	R_c	coil resistance	R_{eq}	equiv. resistance
P	chamber pressure	V	chamber volume	Ω	section
m	mass transfer	γ	polytrophic exponent	P_{in}	pressure before valve
P_{out}	pressure after valve	T_{in}	temperature before valve	ρ	pressure compression ratio
ε	degree of dead volume				
p	oil pressure	W	oil velocity vector	μ	oil viscosity
ρ	oil density	δ	film thickness	r	radial dimension of piston
z	piston axial dimension	p_1	suction pressure	h	piston height
u	oil velocity along radius				

INTRODUCTION

Due to the increasing demand of high efficiency compressors and the strong diffusion of electronic control devices, it is now important to develop better compressor systems. Linear compressor has been studied by several authors, because of the interesting energy consumption capabilities. The paper presents the development of a new single cylinder electrodynamic compressor from the mathematical simulation to the experimental tests. In a linear compressor the motion of the piston is directly due to the magnetic field forces while in standard compressor the rotational motion of the motor is transformed in an alternative motion by a crankshaft mechanism. The linear compressor has a simpler mechanism because the magnetic field impresses an alternative force to the piston. The first advantage is the mechanical simplicity of the system, which can also have a compact and robust structure. The second advantage is the reduction of the friction forces virtually to zero: all forces are directed along the piston motion direction, so there are lower power losses, theoretically there is no necessity of lubrication and there is no wear. The third advantage of this compressor is due to the possibility of varying the capacity by controlling the displacement of the piston. So this compressor can be used at a constant frequency of the current and can have a variable piston displacement. Anyway the possibility presents a disadvantage: controls for the system can be necessary in order evaluate piston position.

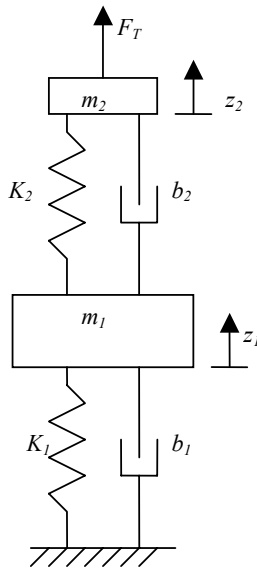


Fig. 1 Mechanical system

MECHANICAL SYSTEM

The mechanical system has been analyzed through a dynamic model developed with Adams software. The advantage of this approach is that it is possible to analyze also flexible elements so modal behavior of these elements at higher frequencies; however it can be also analyzed with a simpler two degree of freedom system (Fig. 1). Defining with the index 1 the characteristics of the stator: (z_1 stator position, K_1 and b_1 suspension spring characteristically stiffness and damping) and with the index 2 the terms related to the piston (z_2 piston position, K_2 and b_2 piston spring characteristically stiffness and damping)

As known, the equations of this system are:

$$\begin{cases} m_1 \ddot{z}_1 + (b_1 + b_2) \dot{z}_1 - b_1 \dot{z}_2 + (K_1 + K_2) z_1 - K_1 z_2 = 0 \\ m_2 \ddot{z}_2 - b_2 (\dot{z}_1 - \dot{z}_2) - K_2 (z_1 - z_2) = F_T \end{cases}$$

The previous differential equation system can be solved numerically.

The external applied force F_T depends on several terms:

1. Magnetic force
2. Pressure force applied by the gas
3. Gravity force
4. Friction force

The first force (F_{el}) is due to the electric system and will be analyzed in the paragraph *Electric system*.

The second force (F_{th}) depends on the thermodynamic cycle and on the valves dynamic. It will be analyzed in the paragraph *thermodynamic system*.

The third term has impact on static equilibrium position, but has no relevant importance on the model. The forth term is due to friction forces along sliding parts. It will be analyzed in the paragraph *friction losses*.

Impact forces on piston and valves have been considered in the model, using Adams features.

ELECTRIC SYSTEM

From Lorenz force equation it is known that magnetic forces can be calculated:

$$F = \oint i \wedge B dl$$

So the oscillating motion is due to the fact that the current is alternative.

In the hypothesis of stationary conditions, the force applied by the electric motor is:

$$F_{el}(t) = B_{eff} \cdot l_{eff} \cdot i(t)$$

Where: B_{eff} is the effective magnetic flux density that acts on the coil, l_{eff} is the effective length of the coil wire. The current $i(t)$ is obtained by solving the equations of the equivalent electric circuit (Fig. 2). For a transient analysis, for example during the start up, or for a correct evaluation of higher harmonics, it is necessary to evaluate correctly the current in the circuit. At first approximation it is possible to consider the motor as a constant resistance and inductance in series. A better model considers a resistance in parallel to the inductance of the motor due to the iron losses. From Kirchhoff equation of circuit:

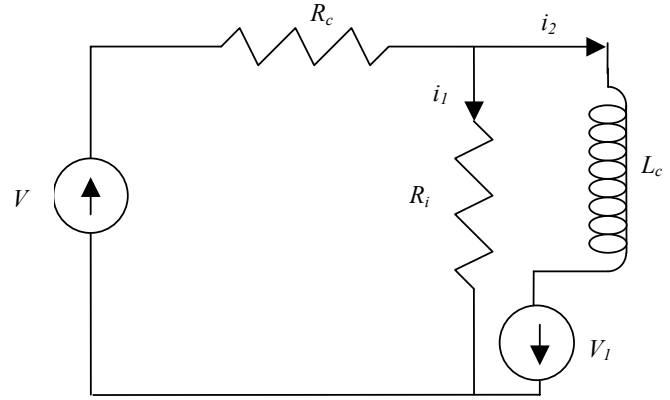


Fig. 1 Electric system

$$\begin{cases} (R_c + R_i)i_1 + R_c i_2 = V \\ R_i i_1 - \frac{dL_c}{dz} \dot{z} - L_c \frac{di_2}{dt} - \frac{d\Phi}{dz} \dot{z} = 0 \end{cases}$$

Where: R_i is the equivalent of iron losses, L_c is the coil inductance and R_c is the coil resistance. Term:

$$V_1 = \left(\frac{dL_c}{dz} + \frac{d\Phi}{dz} \right) \dot{z}$$

is due to the relative motion of the two parts. In the hypothesis of stationary conditions, it is possible to simplify:

$$R_{eq} = R_c + \frac{\omega^2 L_c^2 R_i}{R_i^2 + \omega^2 L_c^2} \text{ and } L_{eq} = \frac{R_i^2 L_c}{R_i^2 + \omega^2 L_c^2}$$

As previously written, this approach can be used in stationary conditions, considering only the first harmonic. It is possible to consider higher harmonics solving the equations with a Fourier series of generators of sinusoidal frequencies also for higher frequencies. It is obvious from the previous equations that higher harmonics will have significantly lower inductance, and for these the circuit will be mainly seen as the coil resistance. Then the equation of the electric circuit is:

$$V = V_1 + R_{eq} i + L_{eq} \frac{di}{dt}$$

where: V is the voltage source. V_1 is due to the relative motion of the stator and magnets. If inductance is constant varying the position, the expression of the induced motional voltage is:

$$V_1(t) = B_{eff} \cdot l_{eff} \cdot (\dot{z}_2 - \dot{z}_1)$$

As known the power consumption of the system is: $W=VI$, so it is possible from the model to evaluate also this parameter and the coefficient of performance of the system.

THERMODYNAMIC SYSTEM

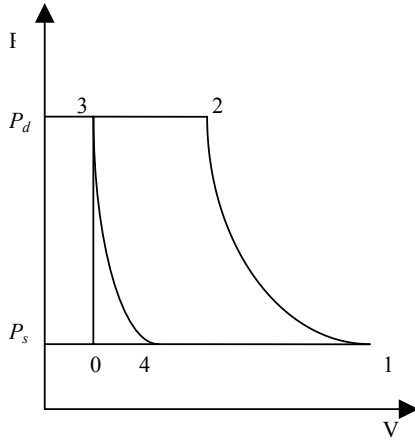


Fig. 2 Ideal thermodynamic cycle

Force applied by the gas on piston (F_{th}) is proportional to the piston pressure, which depends on the thermodynamic cycle. As known the ideal thermodynamic cycle for a compressor (shown of fig. 3) have two polytropic transformation during which there is the state equation's: $PV^\gamma = \text{constant}$

This equation will be used to find the pressure applied on the cylinder during compression and expansion. For a linear compressor piston displacement is not constant, however the equation can be used between two contiguous states.

Suction and discharge can be, in first approximation, considered as isobar transformations. However a better approximation of these transformations considers an adiabatic efflux. In this case from the thermodynamic theory mass flow through a section Ω is:

$$\frac{\dot{m}}{\Omega} = P_{in} \sqrt{\frac{2}{RT_{in}} \cdot \frac{\gamma}{\gamma-1} \left[\left(\frac{P_{out}}{P_{in}} \right)^{\frac{2}{\gamma}} - \left(\frac{P_{out}}{P_{in}} \right)^{\frac{\gamma+1}{\gamma}} \right]}$$

Where the index *in* correspond to the position before the valve. This equation can be used to evaluate mass flow through valves. In this case valves can be approximated as spring – mass systems and in the equation of motion the lift due to the pressure difference $P_{in} - P_{out}$ will be used. Of course the equation of mass flow is important as a control parameter: during the discharge phase mass inside the cylinder can not be less than mass discharged through the valve from the instance it has been opened, and a similar consideration can be done during the suction phase. From the previous equation it is also possible to find maximum of $\frac{\dot{m}}{\Omega}$ varying pressure ratio to avoid high turbulence and flow drop.

It is difficult to minimize dead volume because piston displacement depends on several parameters, as known volumetric efficiency is:

$$\mu = \frac{v_1 - v_4}{v_1 - v_0} = 1 - \varepsilon \left(\rho^{\frac{1}{\gamma}} - 1 \right)$$

where ρ is pressure compression ratio and ε is:

$$\varepsilon = \frac{V_0}{V_1 - V_0}.$$

From the previous equation you can find that:

$$\rho \leq \left(1 + \frac{1}{\varepsilon} \right)^\gamma$$

in order to avoid that $\mu=0$, as in this case the compressor would not pump.

An improvement of the model will consider the heat exchange during the phases. This has clearly an impact on power consumption.

In fig. 4 there is a comparison of pressure volume cycle according to ideal valve and a cycle with simulated spring mass valve. Of course the main difference between the two is due to the consideration of valve dynamics.

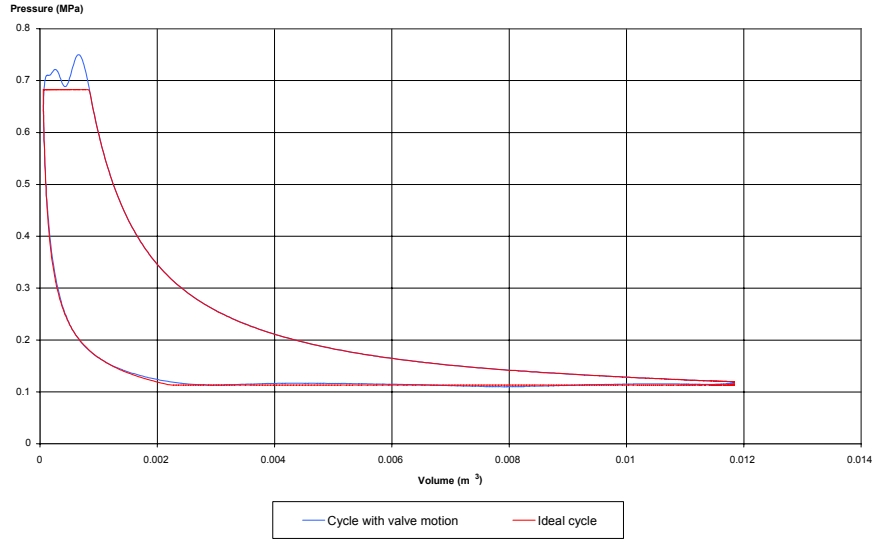


Fig. 3 Pressure volume diagram

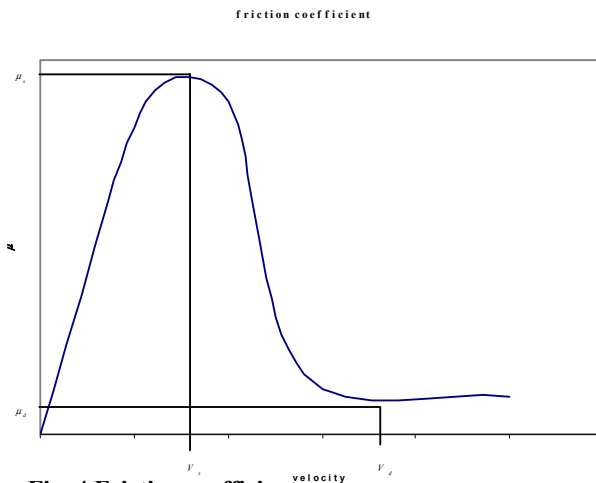


Fig. 4 Friction coefficient

FRICION FORCES

Friction forces have been considered with two different approaches, having the compressor has a lubrication system or not. As previously said, in a linear compressor all forces are directed along piston motion direction. In this case friction forces depend on the mounting of the compressor and gravity could be important. However if there is no lubrication it can be possible to evaluate friction forces due to normal forces applied between the piston and stator and the relative motion by Coulomb friction theory (depending on relative velocity

friction coefficient changes from static to dynamic value).

If there is a lubrication system friction forces can be evaluated from Navier-Stokes equations with the hypothesis of constant viscosity and incompressible fluid:

$$\rho \frac{DW}{Dt} = -\nabla p + \mu \nabla^2 W$$

In the volume between cylinder and piston, considering single directional motion of the system and that oil film thickness δ and velocity are constant:

$$-\frac{dp}{dz} + \mu \frac{\partial^2 u}{\partial r^2} = 0$$

Where: P is the pressure of oil film u is oil velocity, μ and ρ are respect viscosity and density of oil.

Integrating the equation along radial direction r and imposing boundary conditions along radial direction:

$$u(0)=0; u(\delta) = (\dot{z}_2 - \dot{z}_1):$$

$$u = \frac{(\dot{z}_2 - \dot{z}_1)r}{\delta} - \frac{\delta r}{2\mu} \left(1 - \frac{r}{\delta}\right) \frac{dp}{dz}$$

From fluid constitution equation: $\tau_{rz} = \mu \frac{\partial u}{\partial r}$, considering a linear pressure distribution:

$$\frac{dp}{dz} = \frac{p - p_1}{h}$$

where: p_1 is the suction pressure, h is height of the piston. Then friction force on cylinder surface is:

$$F_r = \int_S \tau_{rz}(\delta) dS = \int_S \frac{\mu(\dot{z}_2 - \dot{z}_1)}{\delta} + \frac{\delta(p - p_1)}{2h} dS = \pi\Phi h \left(\frac{\mu}{\delta} (\dot{z}_2 - \dot{z}_1) + \frac{p - p_1}{2} \right)$$

where Φ is piston diameter.

MEASUREMENTS ON THE COMPRESSOR

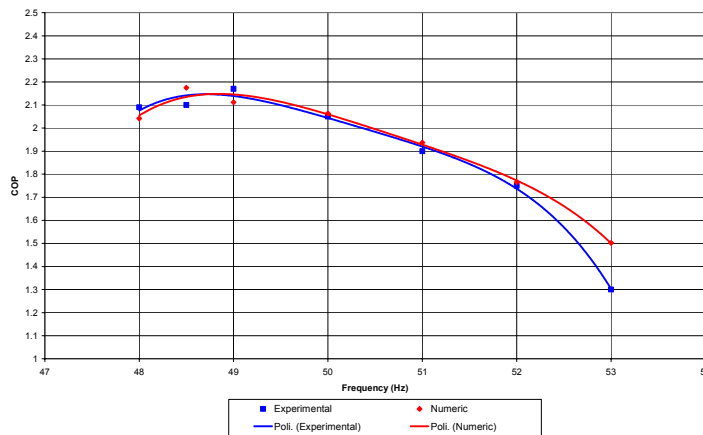


Fig. 5 Efficiency varying current frequency

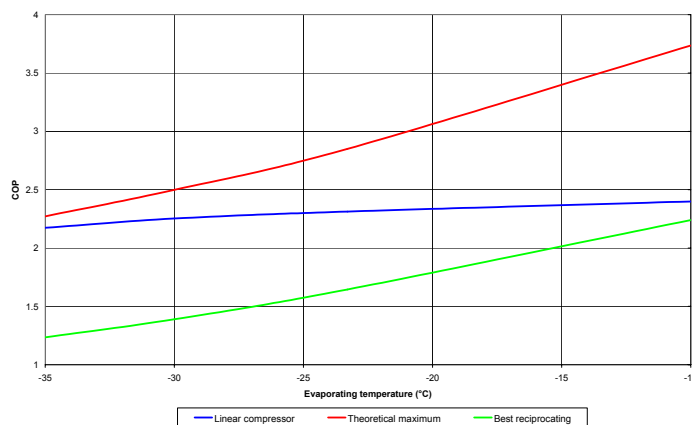


Fig. 6 Efficiency Vs. evaporating temperature

First some measurements on the compressor have been performed to validate the mechanical and electrical input parameters (mass, spring stiffness, damping, resistance, inductance, etc.) from which Adams model evaluate dynamical parameters and efficiency of the compressor. Then a second set of measurements as been performed to validate the results of the simulation. On Fig. 6 it is possible to see the influence of frequency on efficiency of the compressor for the same condition. As consequence it is possible to optimize the system for a given current frequency, but varying the frequency the efficiency will get worse.

Linear compressor has higher efficiency than traditional reciprocating compressor due to lower friction losses. As it is possible to see in Fig. 7, the compressor efficiency is high especially at lower evaporating temperature.

CONCLUSIONS

As conclusion of this study the following considerations can be made:

1. Linear compressor have higher efficiency then standard reciprocating compressor.
2. The system has high sensitivity to the driving frequency, for which it is to be designed
3. Model has a good agreement with experimental results
4. The applied voltage has to be controlled in order to optimize the efficiency of the compressor, with a good knowledge of input parameters.
5. Further optimization will be performed to improve the knowledge on this system

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